FLUID FILM BEARING DESIGN CONSIDERATIONS

by

Stanley Abramovitz

Abramovitz Associates, Inc.
New York, New York

A consultant and manufacturer in the fluid film bearing field, Stanley Abramovitz has, for over 20 years, been intimately involved in the design, development, and production of fluid film bearings for a wide variety of applications.

In 1965 he established Abramovitz Associates, Inc., which specializes in consulting, and the design and manufacture of special fluid bearings. Previously, in 1956, he founded the Continental Bearing Research Corporation, which became a jointly owned division of Industrial Tectonics, Inc., in 1962. As the director of that division, Mr. Abramovitz oversaw a staff that was responsible for development, manufacture and sales of bearings and special products. Earlier, in 1950, he formed and directed the Friction and Lubrication Section of the Franklin Institute.

He received a B.S. degree in Mechanical Engineering from Drexel Institute of Technology in 1948, and an M.S. degree in Mechanical Engineering from the University of Pennsylvania in 1953.

He is a member of several professional and honorary societies. He has a number of patents and several publications in the area of bearing design.

ABSTRACT

The basic and practical design concepts of fluid film bearings as applied to turbomachinery will be discussed. The discussion will cover specific applications that are representative of a wide range of conditions that can be expected. This will include the types of bearings that are most indicated for the specific machinery and their operating conditions. Although the basic theoretical criteria will be included, the emphasis will be on "hardware" examples that describe the bearing design for the particular job.

Speed, and type of fluid lubricant and temperature will basically categorize the design applications. Applications will cover speeds ranging as high as 400,000 rpm. The use of lubricants that are the process fluids result in simplicity of over-all design and are, in many cases, quite feasible. The fluid lubricants covered will be, the conventional oils, the low viscosity liquids such as water, Mercury, etc., and gases such as air.

The discussion will give the designer a feel for the type of bearing most applicable, and information to ballpark the size. Additional influencing factors such as alignment, and the types and effect of shaft and bearing instabilities will be discussed.

INTRODUCTION

The paramount requirement for fluid film bearings in turbomachinery is that they be reliable. This is not a subtle observation, but with the new and attractive bearing design approaches, reliability is sometimes a hopeful rather than an assured requirement. Immunity from bearing failure must cover all normal operating conditions, and also, as far as possible, the abnormal emergencies which occasionally arise. In large turbines the bearings are a small part of the total cost, and yet in other types of machinery the bearings can run as high as 30% of the cost of the entire unit. However, in all cases, machinery shut-down due to bearing failure or impending failure is expensive: not only from the standpoint of the equipment itself, but also as to a break in production such as would occur in the chemical or electrical power industry. Reliability for life support is obvious.

It therefore is necessary for designers to follow proven practices with gradual adoption of new concepts in bearing design, materials, and lubricants. This should not be misconstrued as being ultra conservative, with a resultant limit in progress. There are many examples where bearing technology has been extensively proven, but has not been used in a particular type of machine. An example is a ship forced draft blower that has historically used oil lubricated bearings. By eliminating the use of oil, and selecting a lubricant compatible with the steam of the drive turbine, bothersome maintenance problems and potential hazards can be eliminated. Water lubricated bearings would be applicable, and are under consideration. There is now a long history of successful design and performance information for water bearings in similar applications, yet it would be new for this particular machine. This is a case of using proven bearing technology to achieve simplicity and other advantages in a type of machine that has been in existence for a long time. Over-all, this should be considered a conservative and reliable approach.

Modern machines demand higher speeds, loads, and temperatures. Fluid film bearing technology has kept up with these requirements. The past twenty years have shown an almost exponential rise in the amount of research and development in this field: with associated practical and successful applications. Air as a lubricant was successfully used in experiments seventy years ago, and low viscosity liquids, such as gasoline and water, were applied in certain applications many years before that. Now we consider these classes of fluids real and very usable lubricants, and are able to design the bearings for their use with predictable and reliable results. Other factors such as bearing and shaft instabilities are now understood to the point where the bearing analysis includes the stability considerations. At high speed there can be turbulence in the fluid film. We now understand its effect on film thickness, and can analyze and predict the power loss in the turbulent film.

Applications cover such a wide range of equipment and conditions, that it seems appropriate for this presentation to use the type of fluid lubricant as a parameter. The machinery, conditions, and bearing types used with a particular lubricant would then be described. This also emphasizes a philosophy very close to most bearing designers. That is, the use, wherever possible, of the system working or process fluid as the lubricant. Normally, this eliminates auxiliary lubrication systems and difficult sealing problems, and results in a simpler, sometimes safer, and more reliable machine design.

Therefore, this presentation will be concerned with the type bearings, conditions, and applications using a wide range of fluid lubricants. The generic term "fluid" is used because, besides traditional petroleum oil, lubricants cover a wide variety of liquids, semi-liquids, semi-solids, and gases. Some fluids that have been successfully used as lubricants are: Water, Kerosene, Gasoline, Acid (red fuming nitric), Slurry (acid + sand), Liquid refrigerants, Mercury, Molten metals, Gases (carbon dioxide, helium, nitrogen, air), Vapors (steam, metal), Grease. However, for the purpose of this presentation, the fluids will be divided into three basic categories:

OILS, LOW VISCOSITY LIQUIDS, and GASES.

No attempt will be made to go into the theoretical aspects, since this presentation is rather broad, and a theoretical treatment would necessarily be superficial, and may therefore be misleading. Instead, the reader can refer to the bibliography. The references are a specific and selected list which includes text books and technical papers. They provide excellent design analysis information, and a wealth of additional references.

REVIEW OF PRINCIPLES AND GEOMETRIES

Although there is now a relatively good understanding of fluid film bearings, a brief review may be of value to some. There are a number of bearing choices, and the proper selection is often the starting point and ultimate difference between success and failure.

Fluid film lubrication exists when there is a full film of fluid that completely separates the surfaces of the two members that comprise the bearing. The two fundamental types are the Self-Acting or Hydrodynamic type, and the Externally-Pressurized or Hydrostatic type. The bearing geometries are also of two basic types; the flat surface for thrust loading, and the cylindrical surface for radial loading. In addition, there are Hybrid types that combine Hydrostatic and Hydrodynamic, and spherical and conical shapes for combined thrust and radial loading.

The hydrodynamic bearing is the most common type. It is characterized where the pressure in the film is self-induced by the relative motion between the two bearing member surfaces. *Fig. 1a* shows the classic hydrodynamic wedge with complete film separation, and *Fig. 1b* shows a three dimensional profile that is developed and supports the bearing load. The hydrostatic bearing depends on an external source of fluid pressure and flow to separate the surfaces and support load. *Fig. 2* shows a typical hydrostatic bearing system.

Hydrodynamic bearing geometries must be receptive to the formation of a film pressure wedge. The most reliable and sophisticated type of thrust bearing is the tilting-pad (shoe) bearing. The pad rests on a pivot and is free to incline as it pleases, depending on speed, load, and fluid viscosity. A schematic thrust bearing is shown in *Fig. 3a* with a rotating collar and a series of pad segments, each of which would be mounted on pivots. The pads and their pivots can be mounted in a fixed base. However, in many of our modern machines, high speeds and high temperatures can produce mechanical and thermal misalignment which is accentuated in larger size bearings. The surface of the thrust collar would rotate in a plane that would not be parallel to the pad surfaces, or it could have a wobble effect, or both. The equalized thrust bearing in *Fig. 3b*

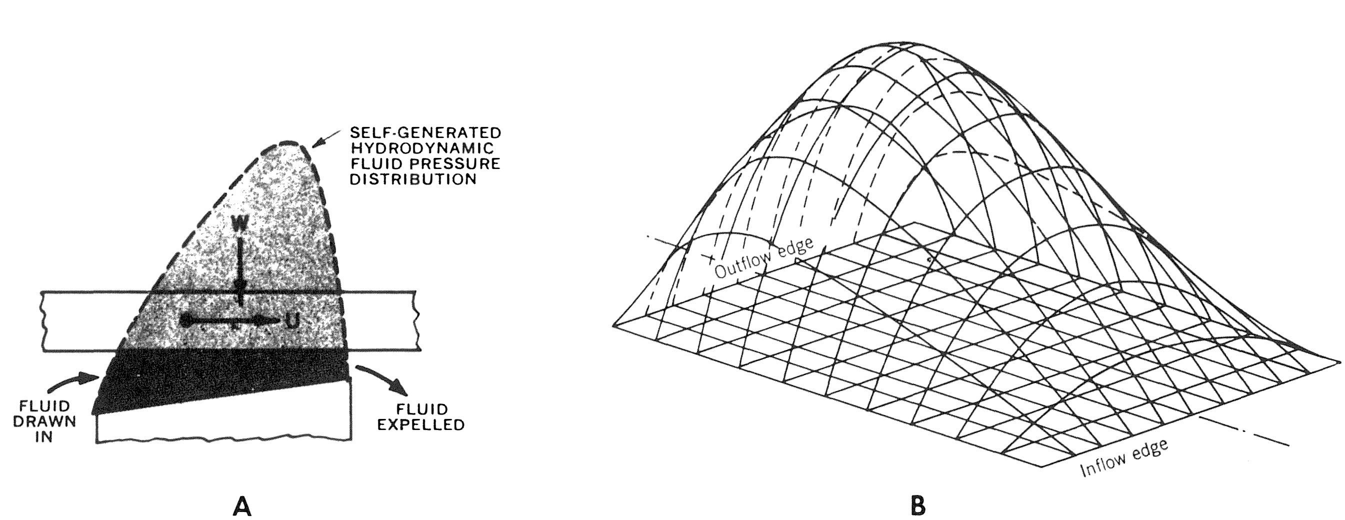

Figure 1. (a) Classic Hydrodynamic Wedge; (b) Three Dimensional Pressure Profile.

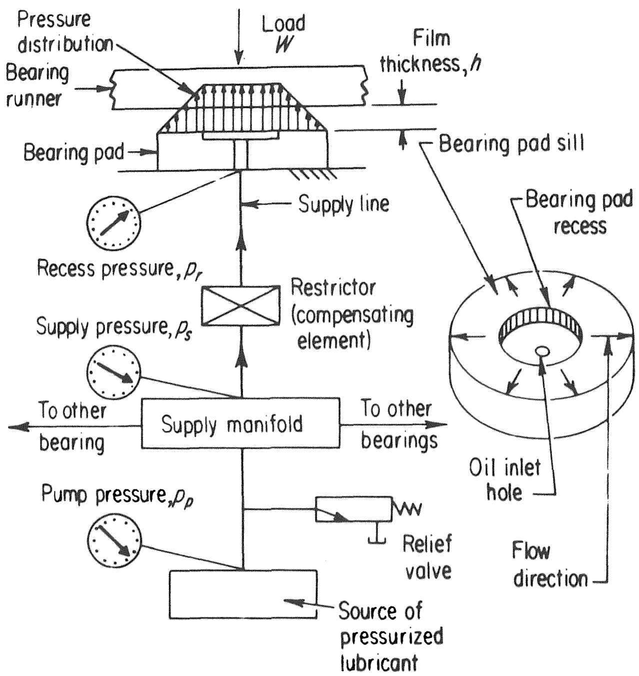

Figure 2. Typical Hydrostatic Bearing System.

has the pads and their pivots mounted on a series of leveling plates which enables each pad to carry an equal share of the load even when there is this misalignment. This is the most expensive type thrust bearing, but has proven to be the solution to many bearing failures where other simpler type geometries have been used. There is an intermediate method which is an unequalized tilting-pad bearing mounted on a spherical seat, *Fig. 3c*. One should be warned that this is effective for initial alignment, and that under load and during operation, the friction at the seat surface may be so high as to render it ineffective. There are other types of thrust bearing geometries in fairly wide use; the Tapered-Land bearing, *Fig. 4a,* and the Step bearing, *Fig. 4b,* where the inclined surfaces or steps are machined on the stationary member. The inclination or step is of the same magnitude as the film thickness, and is determined by a specific set of operating conditions. Any deviation from these operating conditions would reduce the effectiveness of the bearing. In addition, there is the flat land bearing which is a flat plate divided into segments by radial grooves. It should be emphasized that the judgement in selecting one of the above can be the major factor for successful operation. There have been many cases where a bearing problem was solved by only going to the more elaborate type with no change in size or pad characteristics.

Some recently developed geometries that are proving quite successful are the Spiral Groove bearing, *Fig. 5a,* where pressure is developed by the spiral steps, and Compliant Surface bearings, *Fig. 5b* (shown as a journal bearing), where the stationary element is either a thin material supported so it can deform freely, or a resilient material that can predictably distort to a shape that will support load and take misalignment. In addition, bearing surface profiles such as spherical and cylindrical convex surfaces are conducive to film formation and have been used for particular effects.

The most common type of journal bearing is the straight sleeve bearing, *Fig. 6a,* where the diametral clearance between shaft and bearing gives a difference in curvature and an effective hydrodynamic wedge. Sleeve bearings have a wide variety of grooving and shapes. Tilting-pad journal bearings, *Fig. 6b,* are finding more and more usage because of their freedom from self-generating instabilities. They can also be self aligning by having their pads on spherical pivots. Tapered Land, and Step types are also used for radial applications by machining the configuration into a cylindrical bore.

The hydrostatic type bearing as described previously is shown as a journal and thrust bearing in *Fig. 7,* but can be used for conical and spherical shapes as well. In addition to an external source of lubricant pressure and flow, other external equipment such as valves, flow restrictors, and good filters are required. This type has the important advantage that the film thickness and load capacity is primarily a function of the fluid pressure and

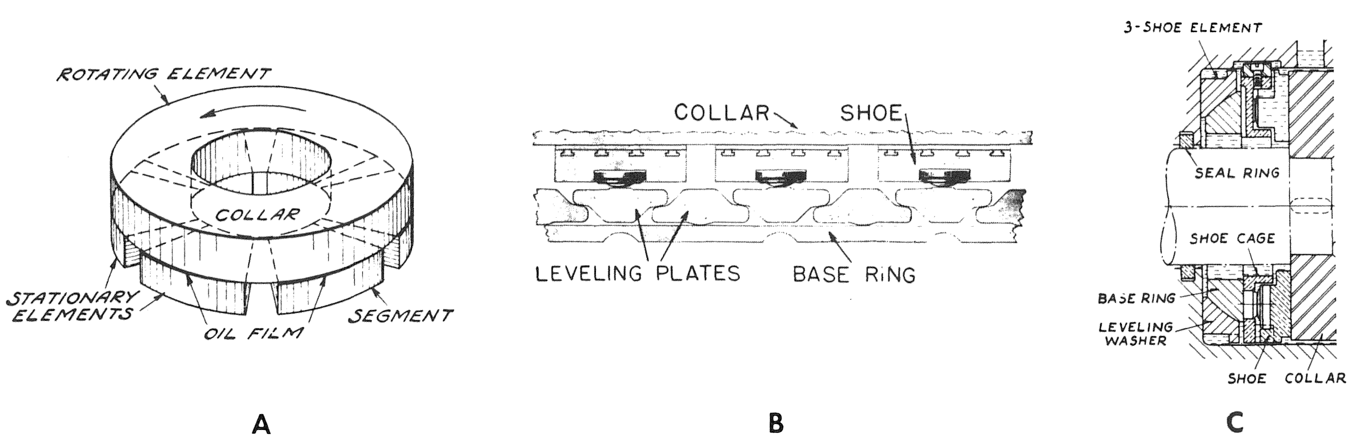

Figure 3. (a) Schematic Thrust Bearing; (b) Equalizer System; (c) Spherical seat arrangement. (Courtesy of Kingsbury Machine Works, Inc.)

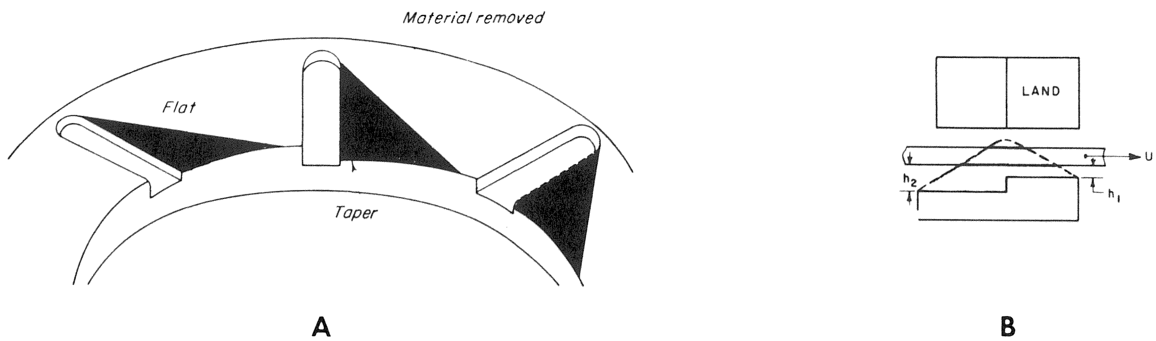

Figure 4. (a) Tapered Land Bearing; (b) Step Bearing.

flow. Therefore the bearing load capacity does not depend on speed. However, the system with its reserve capacity is completely dependent on the reliable operation of the auxiliary components.

The combined use of hydrodynamic and hydrostatic has important application. With heavy dead weight loads, a supply of pressurized fluid may be used during startup or shutdown, and shut off at speed where the bearing operates purely hydrodynamic. In other cases, such as small high speed bearings, the hybrid combination is used during bearing operation to give additional capacity, and in some instances, to eliminate instabilities.

BEARING AND SHAFT INSTABILITIES

One of the most serious forms of instability encountered in journal bearing operation is known as "Half Frequency Whirl." This is one of self-excited vibration and is characterized by having the center of the shaft orbit around the center of the bearing at a frequency of approximately half of the rotational speed of the shaft, *Fig. 8*. The shaft system may be stable, as the speed is increased, until the "whirl" threshold is reached. When the threshold speed is reached the bearing becomes unstable, and further increase in speed produces a more violent instability until eventual seizure results. Unlike an ordinary critical speed, the shaft cannot "pass through" this one and the instability frequency will increase and follow that half ratio as the shaft speed is increased. This type of instability is associated primarily with high speed, lightly loaded bearings. At present, this form of instability is well understood and can be theoretically predicted with good accuracy, and then avoided by altering the bearing design. It should be noted that the tilting-pad journal bearing is almost completely free from this form of instability. However,

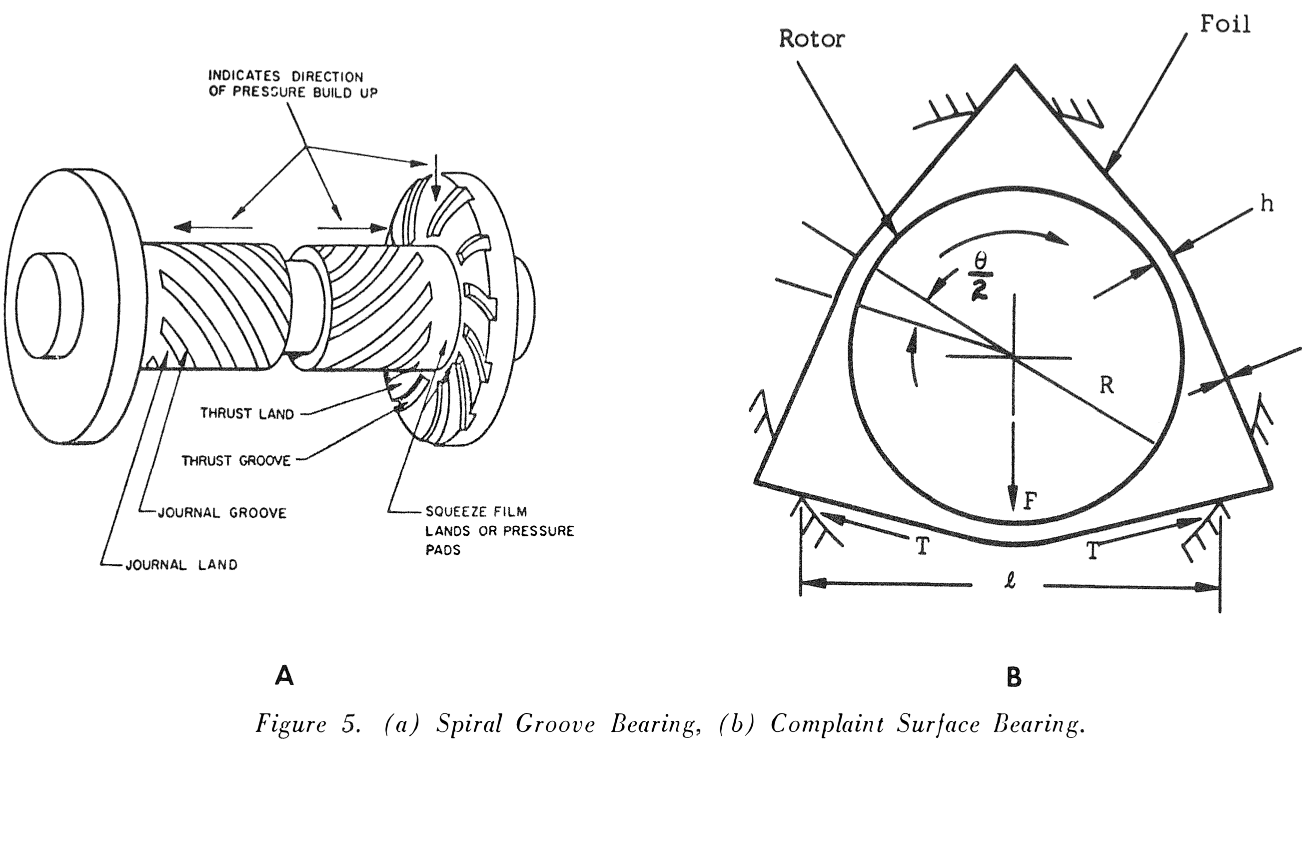

Figure 5. (a) Spiral Groove Bearing, (b) Complaint Surface Bearing.

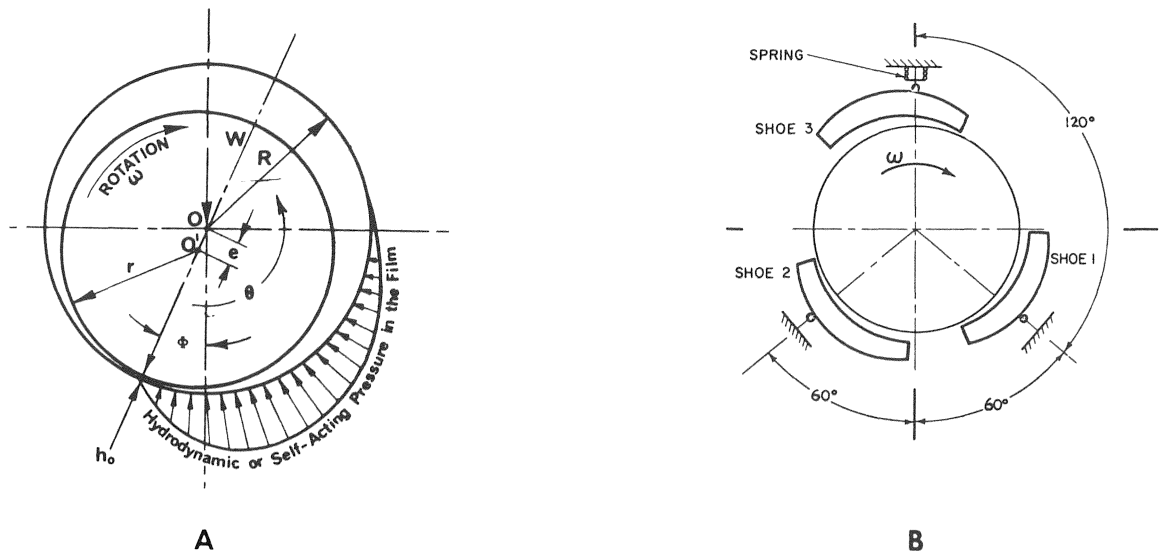

Figure 6. (a) Sleeve Type Journal Bearing; (b) Tilting-pad Journal Bearing.

under certain conditions, the tilting pads themselves can become unstable in the form of shoe (pad) flutter. This is not too common with liquid lubricants, and is most characteristic in gas lubrication because of the low damping of the gas film.

One characteristic that is now being fully utilized is that the bearing fluid film behaves like a spring, that is non-linear. Fundamentally, this involves setting up a curve of load versus film thickness for the particular bearing. The bearing stiffness at any load value can then be obtained from the slope of the tangent to the curve at the load point. *Fig. 9* is an example using a liquid lubricant. It is now common to use film stiffness in determining the critical speed of the rotor, and very complex solutions are obtained using computer techniques. The importance is emphasized by a situation where a unit developed bearing trouble. The shaft bending critical speed had been calculated with the bearings as simple supports (bearing springs infinitely stiff). A recalculation using a realistic value of bearing film stiffness showed that the critical speed was actually one-fourth of that originally predicted. *Fig. 10a* shows, schematically, a rotor shaft supported on two springs that represent the bearings. In most cases journal bearings must carry mass unbalance of the rotating component. This unbalance produces a radial force that rotates at the shaft speed and is a function of the unbalance mass and the rotational speed. Because of the

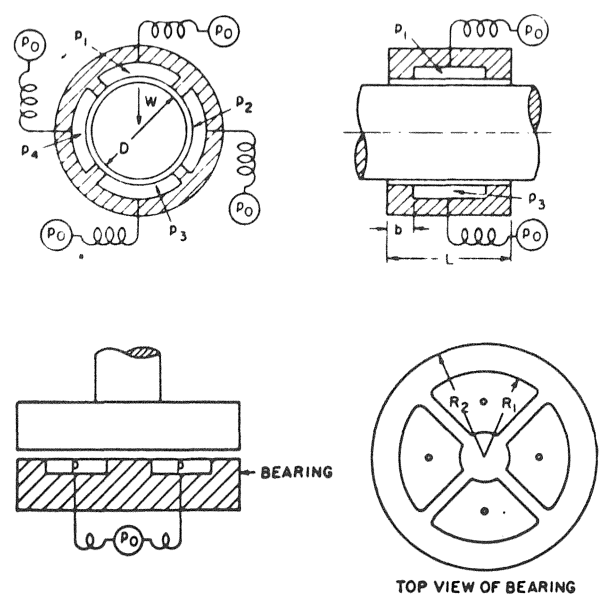

Figure 7. Hydrostatic Journal and Thrust Bearing.

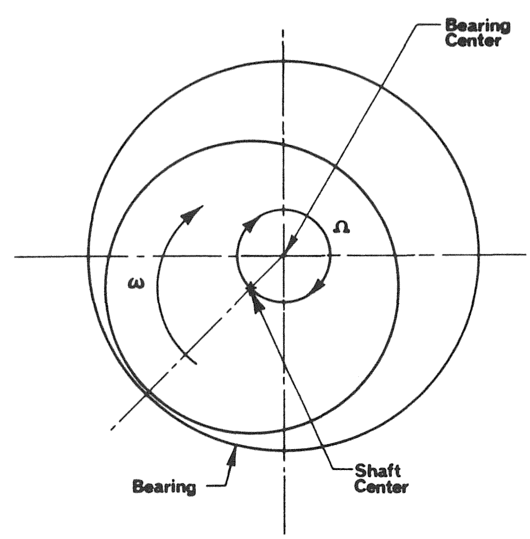

Figure 8. Orbital Whirling of Shaft.

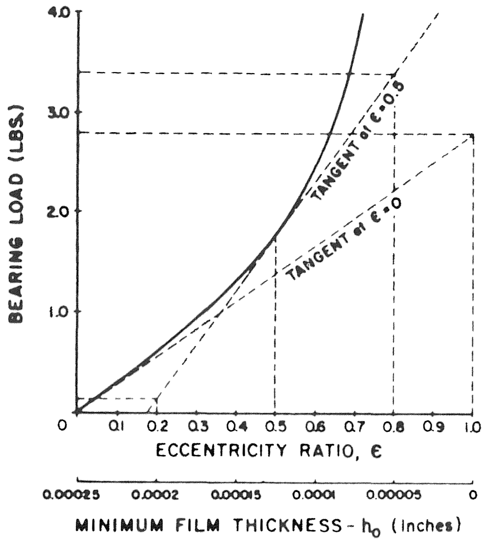

Figure 9. Journal Bearing Load Capacity versus Minimum Film Thickness.

angular motion of the force, the journal will describe a circular orbit in the bearing. Since it is the same frequency as the shaft speed it is generally termed "Synchronous Whirl." *Fig. 10b* shows the shaft motion in "Transilitory (cylindrical)", and "Conical" Synchronous whirl. The number of types of synchronous whirl and combinations with half frequency whirl create problems that are quite complex. However, in the past decade the effort in this area has been so intense that we now have a reliable understanding of these instabilities, and the tools to design bearings accordingly.

With higher speeds and unusual fluid lubricants, Turbulence in the fluid film is no longer rare. We normally think of the thin film as being laminar, but with high speeds, low viscosity and sometimes high density fluids, the lubricant can be turbulent in the film space. This primarily manifests itself as an abnormal increase in power loss. As compared to laminar flow conditions, a Reynolds Number even in the transition region can double the power, and deep in the turbulent regime, can increase the power ten fold. Even though this phenomenon, because of its random nature, is difficult to analyze, there is an unusual amount of theoretical work that has been done, and some experimental work that is available. Just as a guide, one can assume that the transition point will occur at a Reynolds Number of about 800. As to film thickness, there is evidence to indicate that under turbulent conditions it is actually larger than as calculated, based on laminar flow theory.

RELATION OF BEARING DESIGN TO TYPE OF LUBRICANT

Since the primary objective in a fluid film bearing is to maintain separation between the two relatively moving surfaces, the value of film thickness is the basic measure of the bearing load capacity. In hydrodynamic bearings the film thickness is a function of speed, load, bearing size, and fluid viscosity. In hydrostatic bearings it is a function of fluid pressure and flow, bearing size, and fluid viscosity. So with the machine performance and approximate size requirements defined, the fluid viscosity is a critical variable, although this too may be restricted by temperature and power requirements. Bear in mind that although the tendency is usually to strive for a higher viscosity and a more conservative fluid, there are an increasing number of applications where it is advantageous and sometimes mandatory to use a low viscosity liquid where oil had been used, and a gas instead of a liquid.

Oil has been and still is a most reliable fluid lubricant. It has a high range of viscosity, and petroleum oil, with or without additives, has good boundary lubrication properties. It has good service life, and is relatively free from any tendency to attack materials. How-

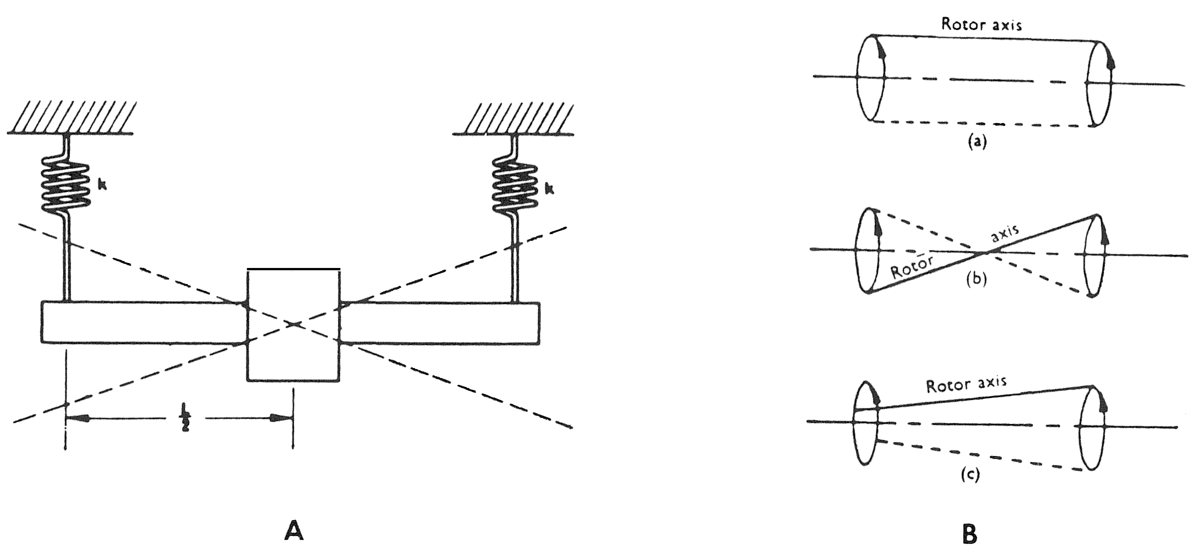

Figure 10. (a) Schematic of Shaft on Fluid Film Bearing Springs; (b) Modes of whirl.

ever, petroleum oil does have a relatively low temperature limit, produces a high power loss even at moderate speeds, and there is a fire risk in machines such as steam turbines. Synthetic oils have been developed that are fire resistant, and are effective at elevated temperatures, but they are very costly and normally have poor boundary lubrication properties.

LOW VISCOSITY LIQUIDS, which are normally the process working fluids, can be from 10 to 100 times lower in viscosity as compared to oil. They almost always have a lower load capacity than oil, but because of the associated lower power loss, higher speeds can be used and reliable values of film thickness developed. Also, in some liquids such as water, higher specific heats reduce the temperature rise through the bearing. Aside from low viscosity, many liquids in this class are corrosive and have poor boundary lubrication properties. In addition, some, with high density such as Mercury and Liquid Metal, become turbulent at relatively low speed. It is therefore necessary to use bearing materials with good boundary properties and corrosion resistance. Since the film may be small, it is very important to provide for misalignment, good filtration, and surface finish and contour. It is felt that this class of fluids provide an excellent opportunity for new bearing applications. There is extensive practical experience, and many types of machines could be made simpler and more reliable, and would be just as comfortable, using a compatible or working fluid instead of a separate oil system.

GASES have a particular place in high speed and/or high temperature applications, and where low power loss is required. The very low viscosity does limit the load capacity for hydrodynamic operation, but hydrostatic gas bearings with high loads are in wide use. The use of gas lubrication has the advantages of simplifying sealing arrangements and eliminating lubricant contamination. Here again, attention to alignment, machining, and bearing materials is very important. Since gases are compressible fluids, the analysis is more rigorous, and, because of the very low damping, instabilities are more prevalent. Gas bearings have not been a laboratory curiosity for quite some time. They can be quite practical, and have proven to be the right solution for many commercial applications.

It is possible to get a "feel" as to whether a particular lubricant is practical for a machine, and also to tentatively size the bearing. The following is a guide for loading and minimum film thickness, as a reasonable design approach. Of course the design must ultimately be analyzed in detail, since in hydrodynamic bearings, speed has a direct influence, and size and anticipated misalignment may dictate thicker films. Unit loading, in psi, is based on the total pad or flat surface area, or the projected area (diameter x length) of a journal bearing.

In hydrodynamic bearings: with Oil—300 to 600 psi, and greater than 0.0010″ film thickness; with Low Viscosity Liquids—50 to 100 psi, and greater than 0.0005″ film thickness; with Gas—5 to 10 psi, and greater than 0.0003 film thickness. In hydrostatic bearings, one can consider as a guide that 1/3 to 1/2 of the available pressure will be usable for bearing load, and also the understanding that the film thickness is a cubic function of the available flow. Comparison curves of film thickness and power loss versus unit loading with the lubricant as the parameter are shown in *Fig. 11*, for hydrodynamic operation. These are for a specific set of conditions, and show the qualitative difference that can be expected in self-acting bearings.

Although this information is useful as a starting point, practical considerations can indicate a more conservative approach. In very large bearings where misalignment and distortions are more pronounced, a 0.0020 minimum oil film may be indicated. Where unequalized

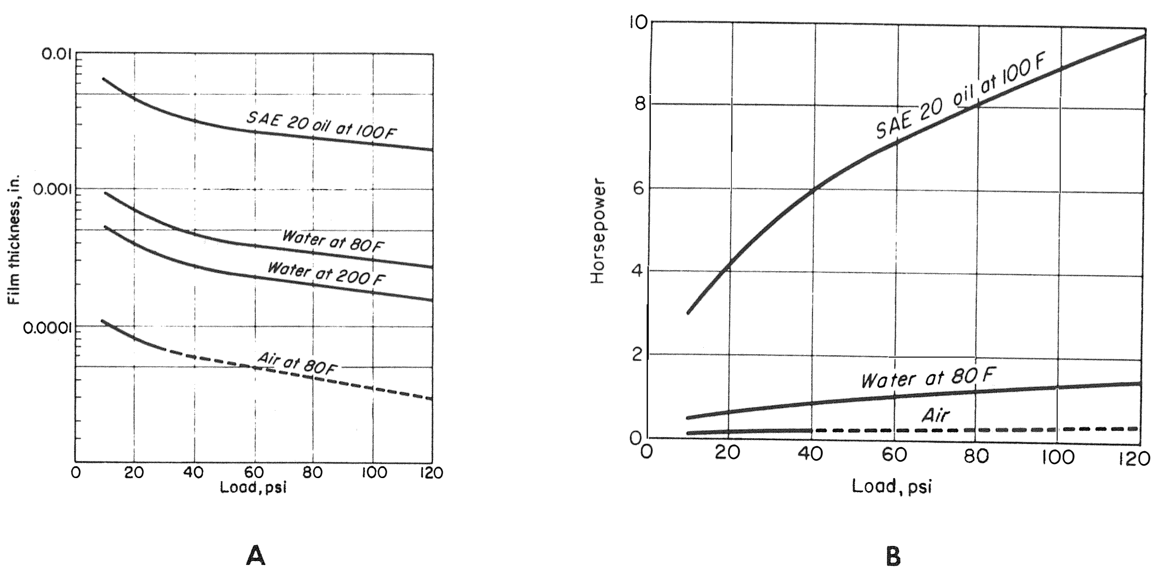

Figure 11. Characteristic Comparison of Bearing Load Versus (a) Film Thickness, and (b) Power, for Various Fluid Lubricants.

thrust bearings are used, it should be assumed that only a portion of the bearing area is supporting the load, and calculations should be based on about three-quarters of the actual bearing area, depending on bearing quality. With a high load and a large film temperature rise there can be severe surface distortions. In tilting-pad bearings the pad becomes convex due to mechanical bending, and thermal bending, where the film temperature is higher than the temperature of the surrounding fluid. Curvature theory does exist, and must be applied in determining bearing performance.

As a final word, the bearing designer must get completely involved with the machine, and not just treat the bearing as a divorced item with a set of operating conditions. A thrust bearing problem can be influenced by the existing journal bearings. Seals and couplings can reflect bearing conditions that may not have been included in the bearing requirements. And vibrations, overloads, unbalance, and thermal and mechanical distortions are some of the things that the machine reflects to the bearing in question.

APPLICATIONS

Oil—Hydrodynamic

Most hydrodynamic oil lubricated bearing applications are not really esoteric, and there are long histories of trouble free performance in a variety of machines. A high viscosity liquid is more forgiving of bearing errors than, say, a gas. But failures do occur. A machine may be upgraded in speed without a bearing redesign, or a bearing type that worked well in the past was used incorrectly in a similar machine with more severe requirements.

As examples of hardware, *Fig. 12* is a 9½" O.D., tilting-pad, equalized thrust bearing shown assembled and split. It is in use in a Steam Turbine, operating at the relatively high speed of 10,000 rpm. The split is for ease of assembly in the turbine, and every pad has a thermocouple for continuous monitoring of film temperature. The normal turbine thrust load was predictable, but there was also a coupling load. The coupling load was due to shaft expansion where the friction in the gear coupling did not allow complete slippage, and the load below the friction force was reflected to the thrust bearing. The maximum value of this added load was somewhat predictable, but the occurrence and actual value was understandably erratic and unpredictable. The bearing was designed for the maximum estimated load. *Fig. 13* is an example of a compact bearing arrangement for a gas turbine, oil lubricated. The tilting-pad equalized thrust bearing takes the major thrust load, the journal bearing is part of the thrust bearing housing, and, to take small reversal loads, a flat faced thrust bearing (not shown) is also a part of the housing opposite the main thrust.

This is an example of selecting the appropriate bearing type. A turboexpander was designed to operate at 20,000 rpm with fluid film bearings, using a MIL 7808 oil. This was a low viscosity fluid since the oil had a lower viscosity curve than SAE 5, and at operating temperature was only about twice the viscosity of room temperature water. It had seemed economical to use a tapered-land thrust bearing, even though a minimum film thickness of 0.0003" was predicted for the operating conditions. The bearing failed, a tilting-pad equalized bearing of the same size was used in its place, and it operated successfully. The rotor and bearing are shown in *Fig. 14*. In this case it was convenient to use hardened and lapped steel pads against a retained Graphitar runner. This combination could take overloads, and stopping and starting without fear of galling or seizure. Babbitted pads against a steel runner could have been used, if not for the relatively high temperature.

It must be emphasized that tapered-land thrust bearings are being used successfully in small and large turbines. Shaft misalignment and distortion are usually the cause of bearing failure. However, another factor is present. With a calculated film thickness of 0.0003", the machined tapers in each pad segment must have a

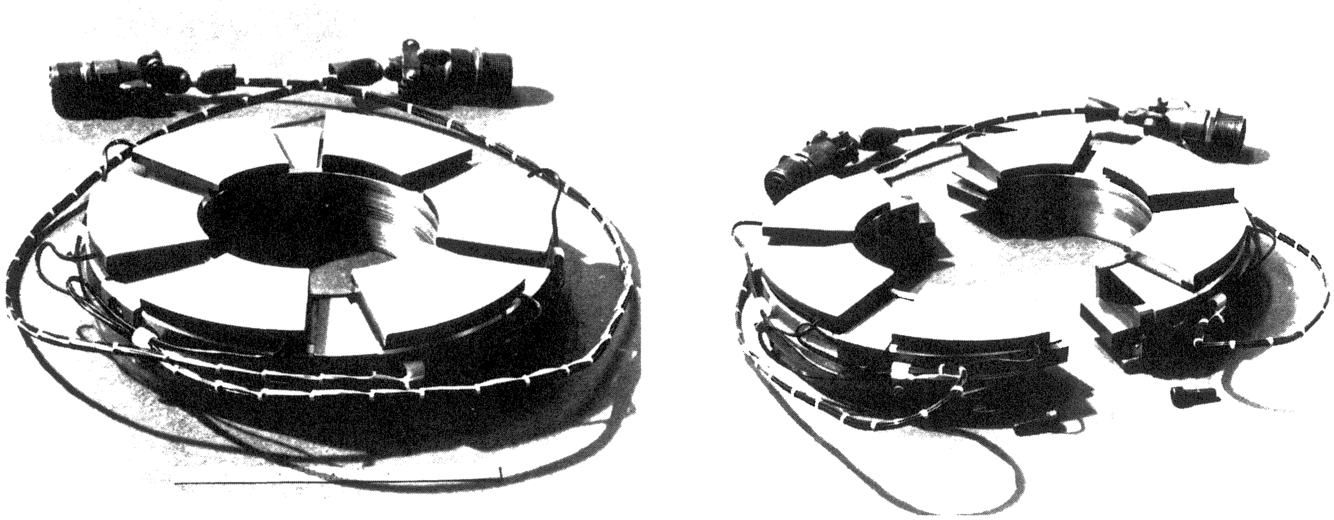

Figure 12. Oil Lubricated Tilting-Pad Equalized Thrust Bearing for Steam Turbine Operating at 10,000 rpm. (designed and built by Abramovitz Associates, Inc.)

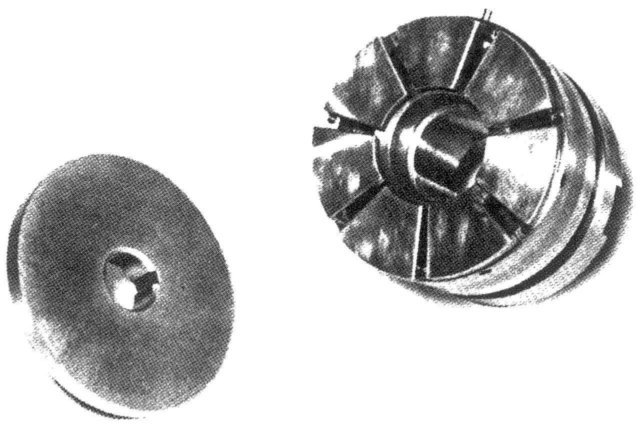

Figure 13. Journal and Double Thrust Bearing Cartridge for Gas Turbine. (designed and built by Abramovitz Associates, Inc.)

drop of about the same value. This is a manufacturing problem, and even holding very close tolerances, the tapers will not be equal, and will either be too large or too small. With this thin film, deviation from the required taper in manufacturing or due to thermal distortion, can be a cause of failure, Etching techniques have been used, and also controlled plating, for step surfaces in gas bearings. However, in relatively standard applications, it would be wise to use a tapered-land or step bearing when film thicknesses are on the order of 0.0010".

Water Lubrication—Hydrodynamic

About twenty years ago during the development of the pressurized water nuclear reactor, one of the problems was the need for a sealess coolant pump. The sealed or "canned" pump bearings would have to use the water being pumped, as their lubricant. Although not strictly new, extraordinary emphasis was placed upon reliability, because of the radioactive hazard associated with servicing the pump. Years of operation without servicing was not inconceivable if the bearings operated with a full fluid film, and the water was kept free of contamination. However, the ability to undergo many stops and starts without seizure or serious wear, and contact and bulk corrosion were considerations equal to the hydrodynamic aspects.

Fig. 15 shows the centrifugal water pump which stood about six feet high, was operated from 900 rpm to 3600 rpm, and was rated at 1200 kilowatts. The journal bearings, shown, had a bore of about 4" diameter. They were a straight sleeve type with helical pumping grooves, and self-aligning features. The thrust bearing, also shown, was a 10" O.D. tilting-pad equalized type. Unit loading on the bearings was between 25 and 50 psi, and the water temperature at the bearings was about 150F.

With the relatively thin fluid films expected, bearing alignment was extremely important. The machine was large, and manufacturing tolerances and thermal gradients had to be considered. The journal bearing ball and sleeve joint was sensitive, *Fig. 16*, and the thrust bearing equalizer system, *Fig. 17*, used hardened buttons and plates to also obtain maximum sensitivity. Materials were Carbon-Graphite (Graphitar) against a Nitrided hardened Stainless Steel or Cobalt based alloys. Since Graphitar is brittle, it had to be retained as shown. In some applications, the thrust bearing construction can be simplified by retaining a disk of Graphitar as the runner, and using solid metal shoes. For less dramatic conditions, bonded phenolics have been used in place of Graphitar. An additional requirement was that the thrust bearing be capable of operating in both directions of rotation. The convex shoe surface was introduced, and enabled a centrally pivoted pad to operate under these low loads and low film temperatures.

Identical pumps that are even larger are being built every year with the same history of trouble free bearing operation. This application provides the confidence and a good basis for other process liquid lubricant applications. The use of low viscosity corrosive liquids, and the need for minimum maintenance are requirements found in many pumps and turbines.

Liquid Metal Lubrication

Mercury lubricated bearings have been used in Turboalternators for space and land based power systems, *Fig. 18*. Liquid Sodium and Sodium Potassium (NaK) lubricated bearings have been used in liquid metal pumps for nuclear power systems. Admittedly, these are very unusual liquids. However, they possess such severe properties that experience with these fluids may be used for other unusual fluids that may have a wider use.

Mercury is about 1½ times the viscosity of water (both at 70°F), and Liquid Sodium at 700°F is about 1/3 that of 70°F water. This low liquid viscosity is familiar, but in addition, the density and corrosive nature of these liquids are quite high. Generally these are high speed applications. Where loads are light or zero (space), the bearings must be selected and designed to

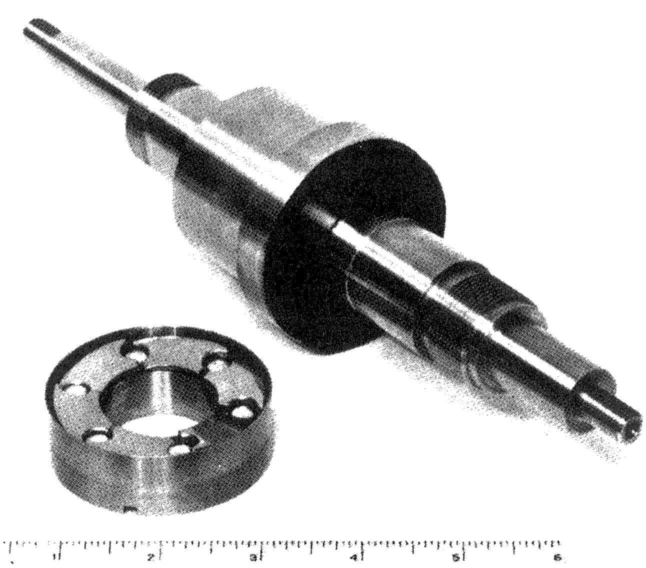

Figure 14. Tilting-Pad Equalized Thrust Bearing and Rotor. (built by Industrial Tectonics, Inc.)

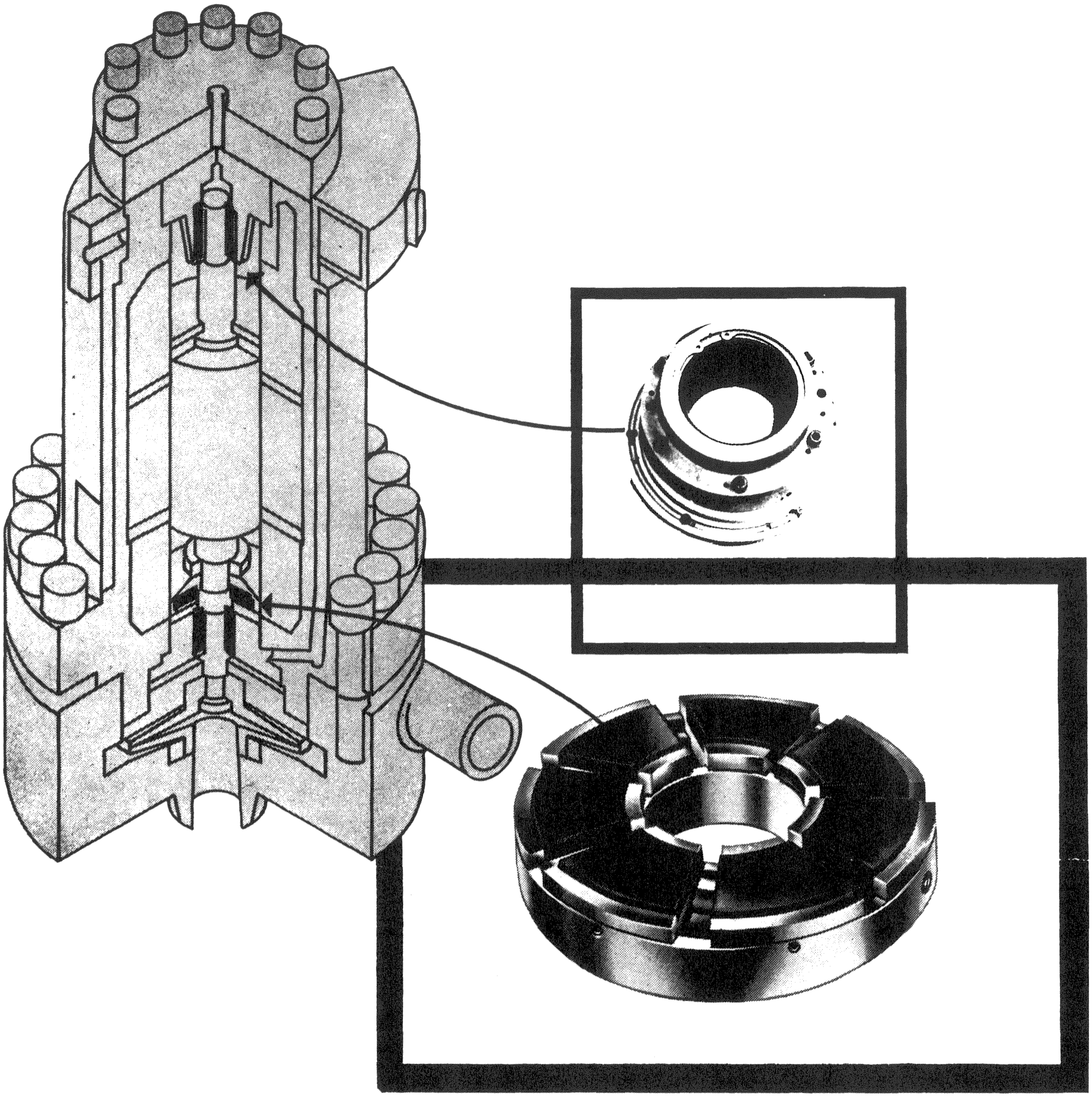

Figure 15. Water Lubricated Journal and Thrust Bearing in "Canned" Motor Pump for Pressurized Water Reactor. (bearings built by Abramovitz Associates, Inc.)

avoid half-frequency whirl instability. Also, the combination of low viscosity, high density, and high speed, places the film deep in the turbulent regime with a resultant high power loss. Bearing materials are critical, and bearing components such as pivots, must be selected for corrosion resistance.

Many bearing types have been used. Journal bearings have been tilting-pad and hybrid pressure fed sleeve types for stability. Thrust bearings have been equalized tilting-pad and spiral groove types. The bearing materials, aside from corrosion resistance, must also have good resistance to wear and seizure, and have low starting friction. As an example, Cobalt based alloys such as Stellite Star J, against a Nickel based alloy, Hastelloy X, was reported to show excellent seizure resistance in Liquid Sodium, although Nickel based materials against themselves were prone to catastrophic seizure. Some Mercury bearings have been made of Carbon Graphite.

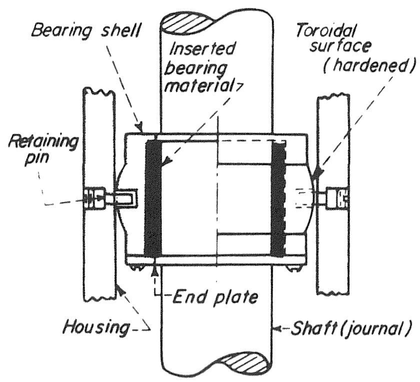

Figure 16. Self Aligning Water Lubricated Journal Bearing with Graphitar Insert.

Hydrostatic Lift—Hydrodynamic Operation

There are situations where a bearing will operate well once there is a reasonable shaft speed, but during starting and stopping, wear or seizure is likely to occur even with good bearing materials. The load will determine the minimum speed at which there is a lift-off and a minimum practical separation between bearing surfaces. That short period of time before lift-off or when stopping can be critical.

Bearings can accept high starting loads where a lubricant, such as oil, has good boundary properties. Where the lubricant offers no help, such as in water and gases, the materials are completely responsible, and lower starting loads are usually required. Some machines produce bearing loads as a function of speed, which offers no starting problem. Although in other machines, either the radical or thrust load includes the shaft assembly dead weight, which may be large. The solution is to apply an external pressure and fluid flow to the bearing surface for starting, and stopping, and cut off the supply when the bearing is at speed and operating hydrodynamically.

Water wheel generators use this arrangement extensively. As an example, a generator with a 64″ diameter tilting-pad thrust bearing was used for peak load operation. The frequency of stopping and starting was high, and were made with high lubricating oil temperatures. The starting conditions were more severe than usual. There was, at times, chatter during starting, and inspection showed high polishing of the Babbitt surfaces. The lifts solved the problem. The arrangement is shown in *Fig. 19.* Each pad surface has an annular groove (F) fed from the manifold (D) through flexible hoses (E). An annular groove is used rather than a recess or small port. This gives a large pressure area and good distribution with the least interference to the normal build up of hydrodynamic pressure when the suit comes up to speed. With a thrust load of 700,000 pounds, a 2000 psi pump was needed to accomplish lift-off, although only about half of that was needed once there was flow. Check valves, filters, and orifices were a part of the auxiliary system. In addition to preventing bearing failure, the lifts also gave very low starting friction.

Another thrust bearing application, using water as the lubricant, was considered for hydrostatic lifts. This was a study of water lubricated bearings for a Helium circulator to be used in a gas cooled nuclear reactor power plant. With high lift-off loads, and other specifications, it was necessary that the external pressure cover a large percentage of the pad area, and yet the alteration of the surface geometry could not interfere with the

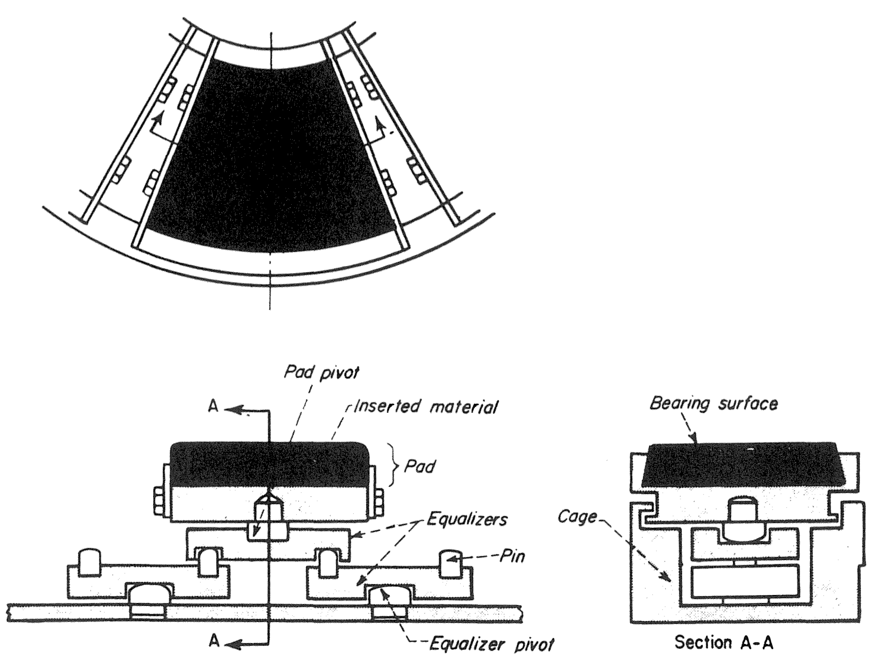

Figure 17. Water Lubricated Tilting-Pad Thrust Bearing with Sensitive Equalizing System, and Graphitar Inserts in Each Pad.

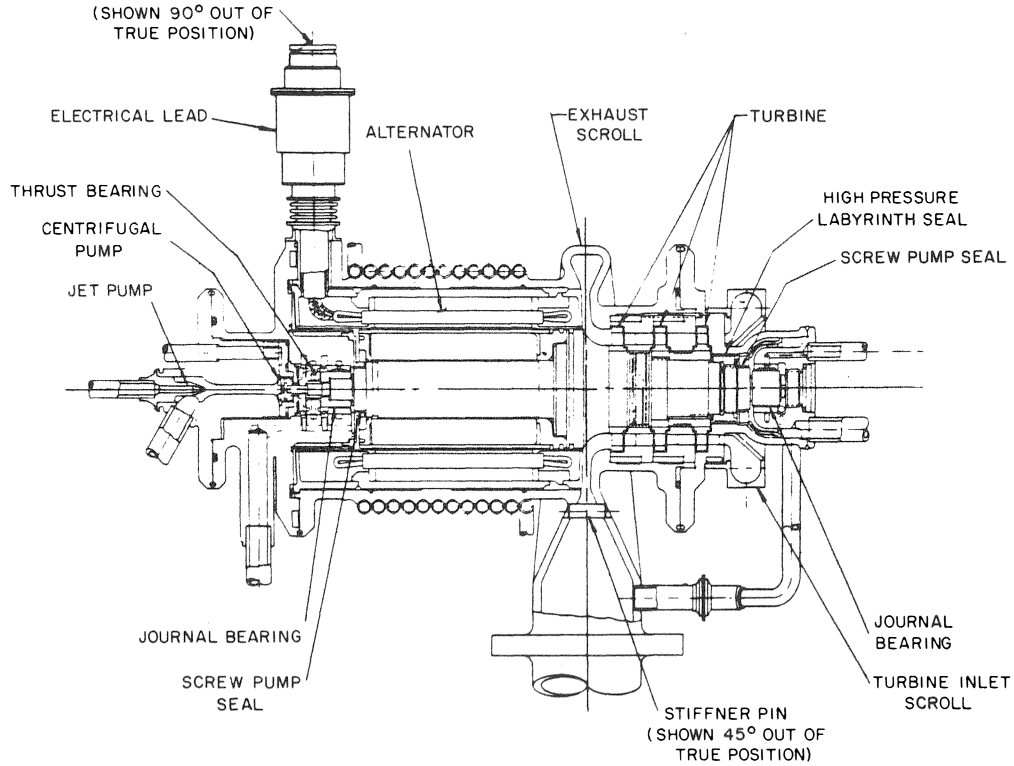

Figure 18. Turboalternator Using Mercury Lubricated Bearings. (developed by TRW, Inc.)

hydrodynamic operation. The bearing design is shown in *Fig. 20.* Five small spot recesses were strategically located on each pad, and check valves were used at particular locations to prevent back flow when hydrodynamic pressure was developed during operation. The external pressure is introduced directly into each pad from a common manifold through tubes that are S-shaped to minimize moments on the pad.

Gas lubricated, tilting-pad journal bearings with hydrostatic lifts were used in a radial-flow turbocompressor. The maximum design speed was 38,500 rpm, and the working fluid was Argon. The bearing pads and arrangement are shown in *Fig. 21.* Four orifices were incorporated near the peripheries of each pad. These provided the lift and eliminated any metal to metal contact at speeds below 20,000 rpm. The hydrodynamic thrust bearing that carried the main thrust load also made use of hydrostatic lifts.

Air Lubricated—Hybrid—Ultra High Speed

The high speed air turbine dental drill was in wide use with ball bearings. At the operating speed of 400,000 rpm and above, bearing noise was uncomfortable and possibly hazardous, and the required oil mist lubrication could also present a health hazard. Since the air pressure source was already available for the turbine, a logical step was to use air bearings. Two designs are shown in *Fig. 22.* One uses straight sleeve journals and thrust shoulders, and the other, a conical shape for combined thrust and radial loading. In both designs the bearings are primarily hydrostatic. Although, with the very high speed, and a relatively tight clearance, some hydrodynamic effect occurs.

Both designs support the bearings on elastic, rubber O-rings. Alignment is one consideration, but some builders have reported that this construction eliminates or reduces the effects of instabilities that are either self-excited or due to unbalance. The rubber allows the

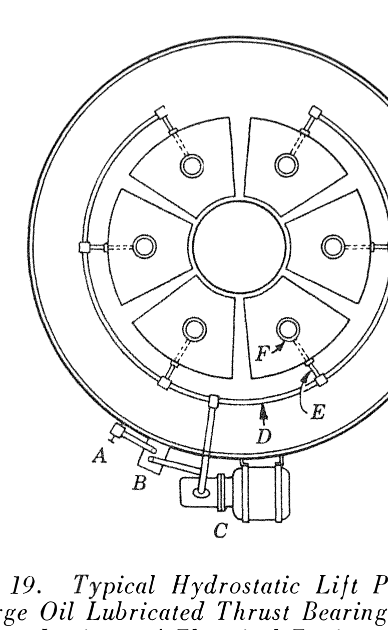

Figure 19. Typical Hydrostatic Lift Piping Diagram for Large Oil Lubricated Thrust Bearing. (courtesy of American Institute of Electrical Engineers).

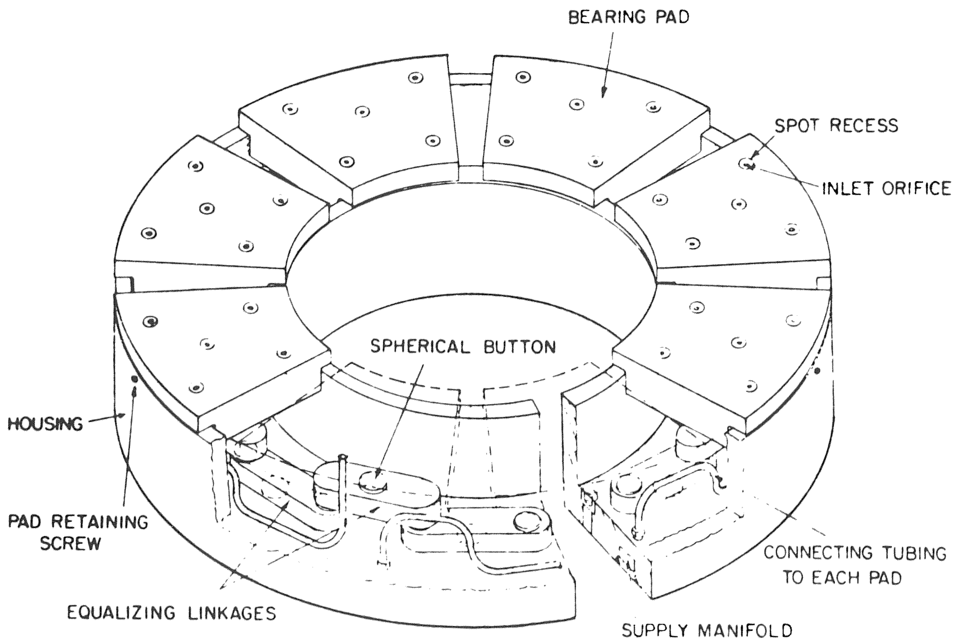

Figure 20. Water Lubricated Tilting-Pad Equalized Thrust Bearing with Hydrostatic Lifts. (developed by The Franklin Institute Laboratories).

bearing to yield and absorb energy from the whirl. Material for the bearing was a very important requirement since the unit is usually sporadically overloaded at high speed. A steel or hard Aluminum rotor has been used against Carbon Graphite, Phenolic, and Meehanite cast iron bearings.

These dental drills are now in wide use and represent a very practical application where the use of air as the lubricant was an ideal solution.

Air Lubricated—Hydrostatic

High speed, turbine driven grinders with air bearings were developed by the machine tool industry. Speeds range from 30,000 rpm to 120,000 rpm. A grinder and bearing arrangement are shown in *Fig. 23*. Normal shop air operates the turbine, and also supplies the hydrostatic bearings. The journal bearings are $1\frac{1}{8}''$ diameter, with a diametral clearance of 0.0010″. At high speed the friction is so low that the temperature rise is only a few degrees, and no cooling is necessary. Required bearing flow is only 3 cfm, and maintenance is nil when there is proper air filtration using a 5 micron filter.

There are also commercial machine tool spindles that use hydrostatic oil lubricated bearings. The stiffer film assures stability and rigidity of the spindle, and has a direct influence on surface finish and roundness of the part being machined.

Helium Lubricated—Hydrodynamic—High Temperature

This example describes a self-acting gas bearing application, and in addition, two approaches: one, a horizontal machine with the emphasis on the journal bearings, and the other, a vertical machine where the thrust bearing carries the major load. The application is a gas circulator to circulate Helium at 1000°F with an ambient pressure of 500 psia. This was a nuclear application, and Helium lubricated bearings would avoid any problems of contamination.

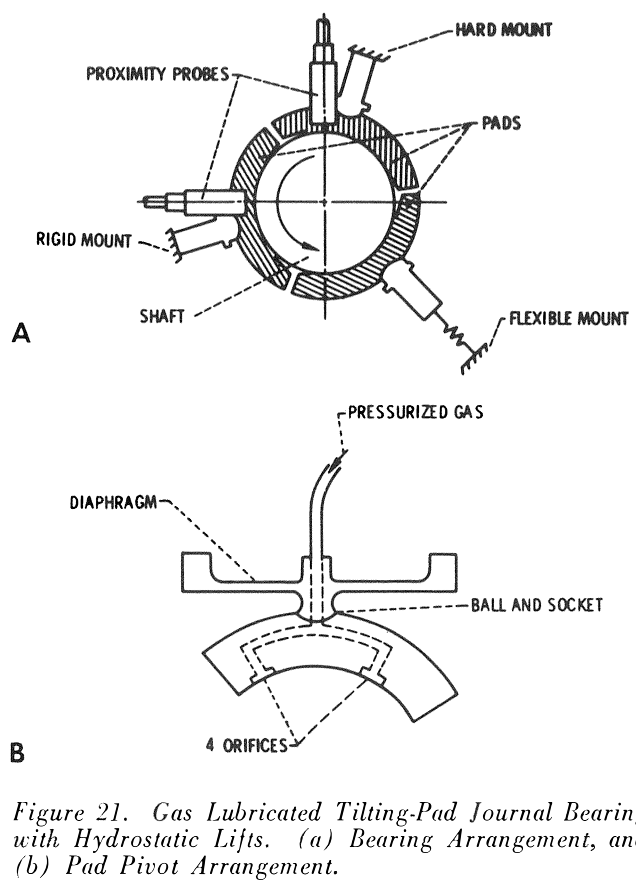

Figure 21. Gas Lubricated Tilting-Pad Journal Bearing with Hydrostatic Lifts. (a) Bearing Arrangement, and (b) Pad Pivot Arrangement.

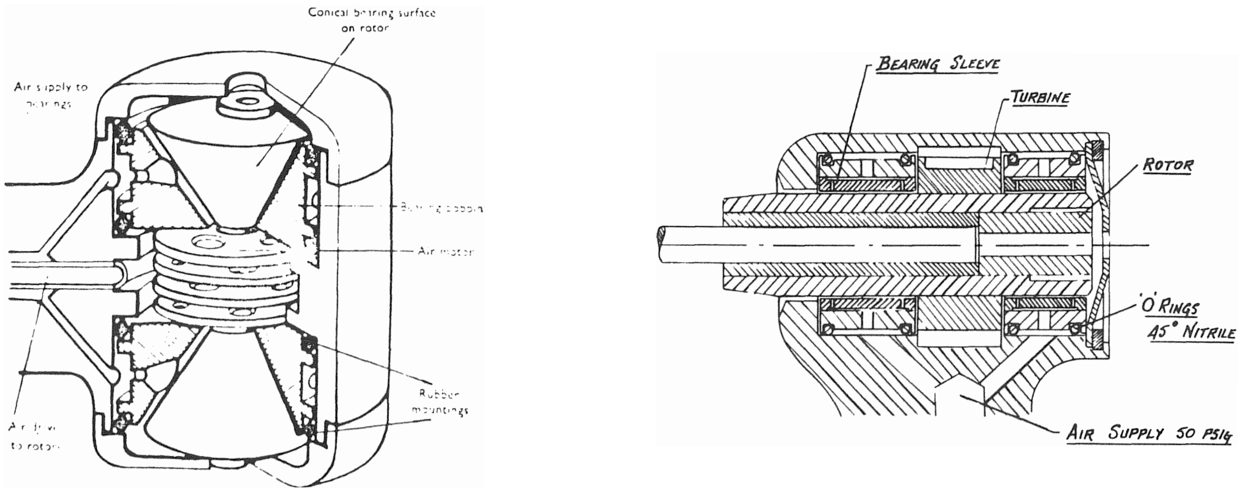

Figure 22. Air Bearing Dental Turbines. (conical shape by Encore Power Division, U.S.A. and straight shape by Westwind Turbines, Ltd., England).

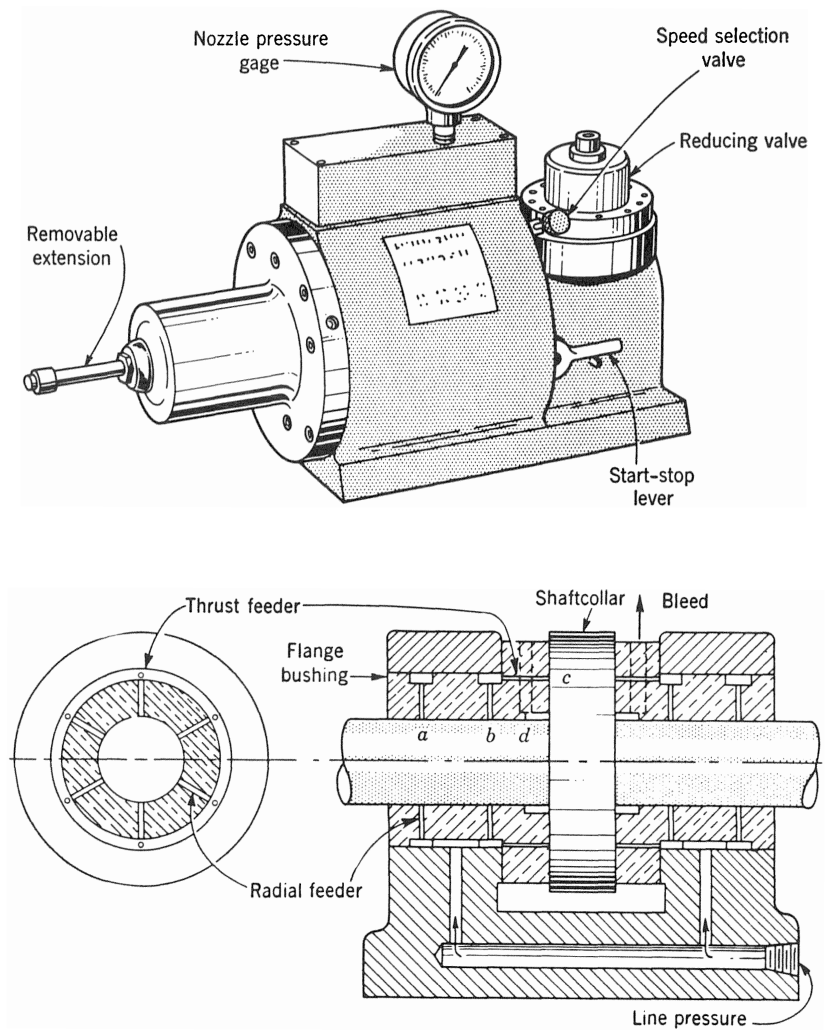

Figure 23. Air Lubricated Grinder and Bearing Arrangement.

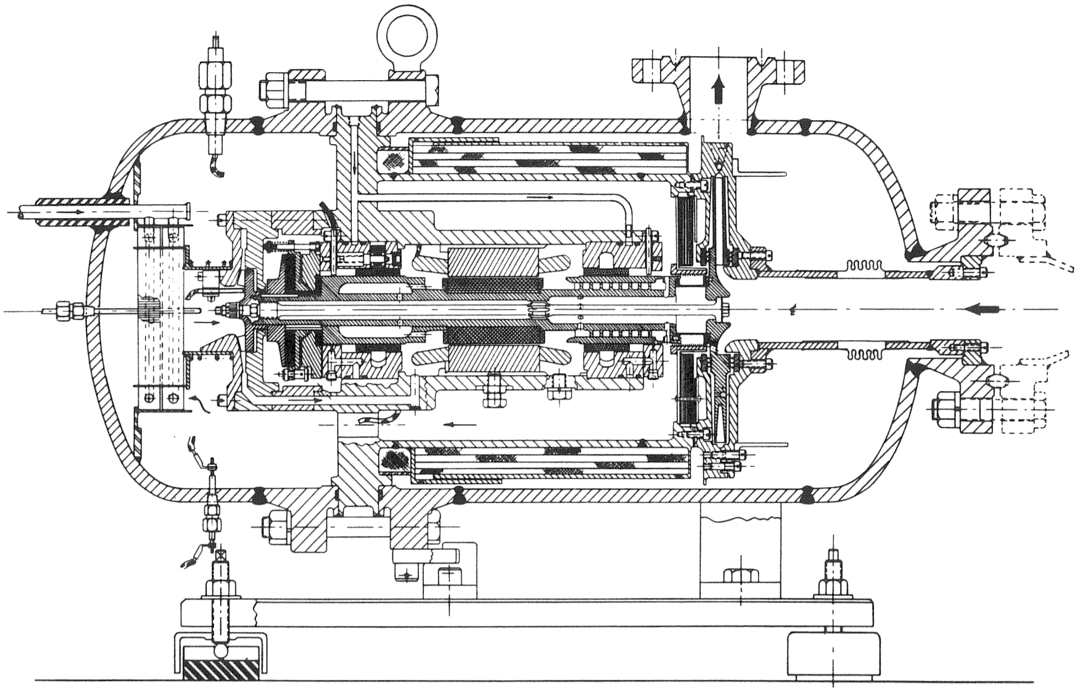

Figure 24. Helium Circulator. (developed by Mechanical Technology, Inc.)

The horizontally mounted circulator is shown in *Fig. 24.* The operating speed is 18,000 rpm. The journal bearings are the tilting-pad type, and the thrust bearing is a self-aligning, helical-grooved plate. As a part of the unit, an auxiliary fan drives Helium gas through an integral, water cooled heat exchanger to cool the motor and bearing areas.

A prototype of a vertically mounted circulator is shown in *Fig. 25.* The journal bearings were the tilting-pad type. They were primarily lightly loaded guide bearings with rather large diametral clearances. The high speed and light load required that type to avoid half-frequency whirl. The self-aligning feature was also a needed advantage. The thrust bearing which took the major load was an equalized, tilting-pad type, with convex surfaced pads. The bearings are shown in *Fig. 26.*

CONCLUSIONS

This presentation described the basic fluid film bearing principles and types, and their use and application with a range of fluid lubricants. It cannot be over emphasized that many bearing problems are created by simply using the wrong type of bearing for the application, or the wrong fluid lubricant for the operating conditions, regardless of how attractive it may seem. The design applications were selected as indicative examples, in an attempt to cover the range of bearing and fluid types.

This information can provide a basis for determining the feasibility of a specific bearing application in comparable machines. At the least, it can give the machine designer the understanding needed to just consider an unusual bearing approach. He can then proceed with the design through the literature, or outside help. The bibliography describes selected reference material, which if still incomplete, can direct the reader to additional and more specific references.

SPECIFIC AND SELECTED REFERENCES

Fuller, D. D.: Theory and Practice of Lubrication for Engineers. John Wiley & Sons, Inc., 1956.

Wilcock, D. E. S.; and Booser, E. R.: Bearing Design and Application. McGraw-Hill Book Company, Inc. 1957.

Pinkus, O.; and Sternlicht, B.: Theory of Hydrodynamic Lubrication. McGraw-Hill Book Company, Inc., 1961.

Gross, W. A.: Gas Film Lubrication. John Wiley & Sons, Inc., 1962.

Grassam, N. S.; and Powell, J. W., eds.: Gas Lubricated Bearings. Butterworths, London, 1964.

Rippel, H. C.: Cast Bronze Hydrostatic Bearing Design Manual. Second ed., Cast Bronze Bearing Institute, Cleveland, Ohio, Apr. 1965.

Standard Handbook of Lubrication Engineering. McGraw-Hill, Inc., 1968.

Bearing and Seal Design in Nuclear Power Machinery. American Society of Mechanical Engineers publication, 1967.

NASA Contributions to Fluid-Film Lubrication—A Survey. Prepared by The Franklin Institute Research Laboratories, NASA SP-5058, 1969.

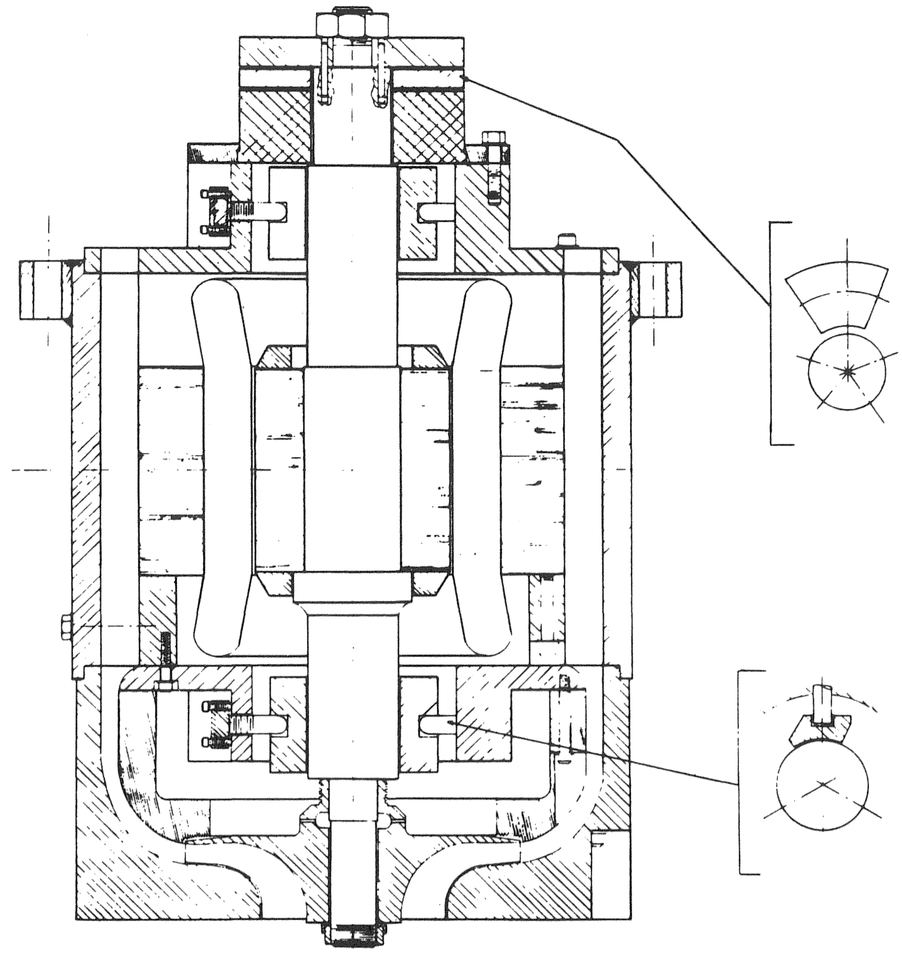

Figure 25. Prototype of Helium Circulator. (developed by Abramovitz Associates, Inc.)

Design Manual for Self Acting Plain Cylindrical Gas Journal Bearings and Thrust Bearings for Use in Electrical Blower Motors. Rotron Manufacturing Co., Inc., Woodstock, New York, 1963.

New Concepts in Bearing Designs and Applications. Industrial Tectonics, Inc., Compton, Calif., 1965.

Fuller, D. D.: A Review of the State-of-the Art for the Design of Self-Acting Gas-Lubricated Bearings. ASME Paper 68-LubS-10, 1968.

Smith, M. I.; and Fuller, D. D.: Journal Bearing Operation at Super-Laminar Speeds. Trans. ASME, vol. 78, no. 3, Apr. 1956, pp. 469-474.

Decker, O.: Design Applications of Fluid Film and Sliding Surface Bearings. Presented at the ASME Design Engineering Conference, May 17-20, 1965, New York City.

Decker, O.; and Reed, H.: Development of Long Life Liquid Mercury Lubricated Bearings for the Sunflower Rankine Cycle Power System. SAE Paper No. 871-D, April 1964.

Cherubim, J.: Solutions of Problems Associated With the Application of Gas-Lubricated Bearings. Presented at the ASME Spring Lubrication Symposium, Boston, Mass., June 4, 1963.

Cheribum, J.; and Abramovitz, S.: A High Temperature Motor Test Unit Utilizing Gas Bearings. Proceedings of meeting at Oak Ridge National Laboratories, U. S. AEC, Tech. Information Service, Oak Ridge, Tenn., April 2-4, 1962.

Abramovitz, S.: Theory for a Slider Bearing With a Convex Pad Surface; Side Flow Neglected. Journal of the Franklin Institute, 259, No. 3, March 1955.

Abramovitz, S.: Turbulence in a Tilting Pad Thrust Bearing. Trans. ASME Vol. 78, Jan. 1956, pp. 7-11.

Abramovitz, S.: Water-Lubricated Fluid Film Bearings are Practical. Product Engineering Magazine, Sept. 30, 1957.

Gunter, E. J.: Dynamic Stability of Rotor-Bearing Systems, NASA SP-113, 1966.

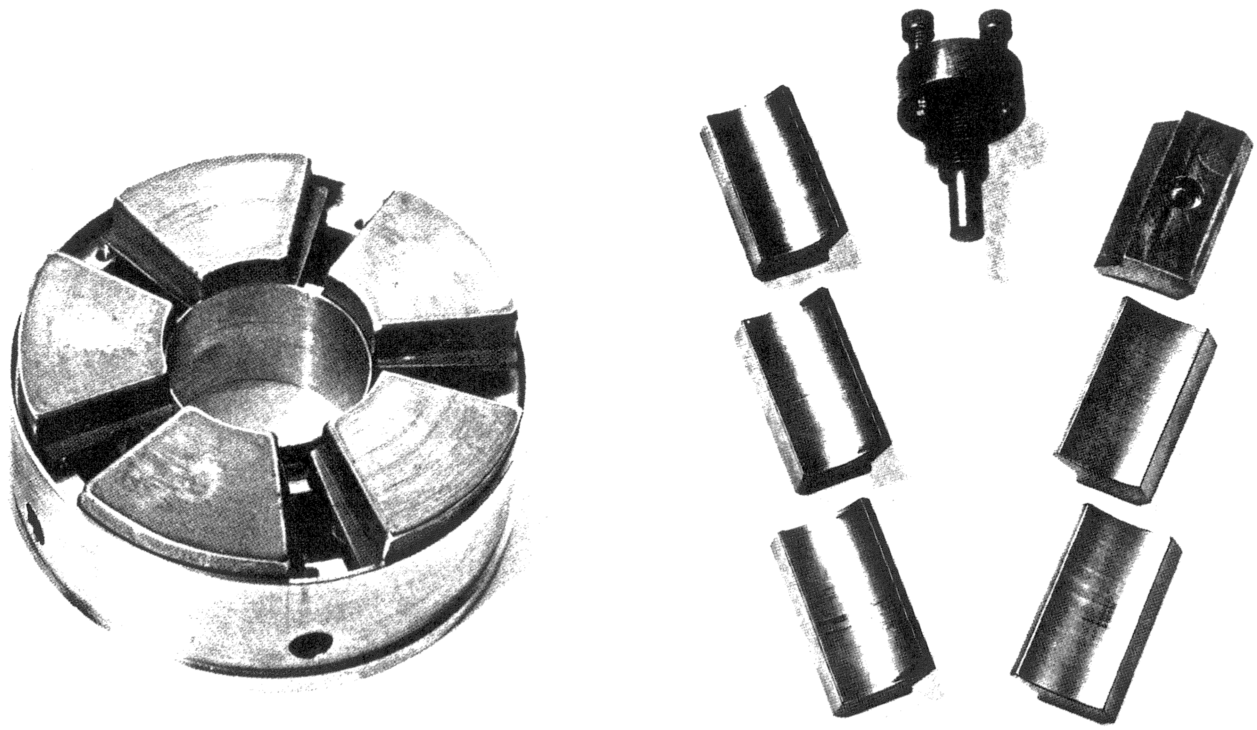

Figure 26. Thrust Bearing and Journal Bearing Shoes for the Prototype Helium Circulator. (designed and built by Abramovitz Associates, Inc.)

Rippel, H. C.: Cast Bronze Bearing Design Manual. Second ed., Cast Bronze Bearing Institute, Cleveland, Ohio, Jan. 1965.

Raimondi, A. A.; and Boyd, J.: Applying Bearing Theory to the Analysis and Design of Pad-Type Bearings. Trans. ASME, vol. 77, no. 3, Apr. 1955, pp. 287-309.

Raimondi, A. A.: The Influence of Longitudinal and Transverse Profile on the Load Capacity of Pivoted Pad Bearings. ASLE Trans., Vol. 3, No. 2, 1960, pp. 265-276.

Constantinescu, V. N.: Analysis of Bearings Operating in the Turbulent Regime. Journal of Basic Engineering, March 1962, pp. 139-151.

Morrant, G. A., Powell, J. W.; and Hargreaves, P.: Air Bearings and Their Application to Dental Air Turbine Handpieces. British Dental Journal, vol. 116, June 16, 1964.

Powell, J. W.; and Tempest, M. C.: A Study of High Speed Machines with Rubber Stabilized Air Bearings. ASME paper No. 68-LubS-9, 1968.